International Journal of Engineering Science Invention
ISSN (Online): 2319 – 6734, ISSN (Print): 2319 – 6726
www.ijesi.org Volume 3 Issue 7ǁ July 2014 ǁ PP.05-18

Experimental Investigation of The Effects Of Condenser Heat Exchanger on The Room Air Conditioner's Performance

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ABSTRACT : *Refrigeration is the procedure of achieving and maintaining a temperature below that of the surroundings, the aim being to cool some product or space to the required temperature. One of the most important applications of refrigeration has been the preservation of unpreserved food products by storing them at low temperatures. Refrigeration systems are also used broadly for providing thermal console to human beings by means of air conditioning. Whereas Air Conditioning is referred to the treatment of air so as to all together control its temperature, moisture content, cleanliness, odor and circulation, as required by occupants, a process, or products in the space. The subject of refrigeration and air conditioning has evolved out of human need for food and comfort, and its history dates back to centuries. The history of refrigeration is very fascinating since every aspect of it, the availability of refrigerants, the prime movers and the developments in compressors and the methods of refrigeration all are a part of it. In the present work the authors has investigated the effect of the following by increasing the area of condenser from 1.5 Row to 2.0 Row. By using this apparatus the author has studied the following parameters:*

[1] Effect of FIN tube heat condenser with 1.5 row coil on the cooling of 1.5TR split air conditioner.
[2] Effect of FIN tube heat condenser with 2.0 row coil on the cooling of 1.5TR split air conditioner.
[3] Comparison of Current for 1.5 row and 2 row condenser with respect to time
[4] Comparison of Power for 1.5 row and 2 row condenser with respect to time.
[5] Comparison of Cooling Capacity for 1.5 row and 2 row condenser with respect to time.
[6] Comparison of C.O.P with 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger.
[7] Comparison of Suction (T°) with 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger.
[8] Comparison of Discharge (T°) with 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger.

INDEX TERMS: *Temperature,R22 refrigerant, COP, Power*

I. INTRODUCTION

Refrigeration systems are also used for providing cooling and dehumidification in summer for personal comfort (air conditioning). At present comfort air conditioning is widely used in residences, offices, commercial buildings, air ports, hospitals and in mobile applications such as rail coaches, automobiles.A refrigeration system can also be used as a heat pump, in which the useful output is the high temperature heat rejected at the condenser [4]. Alternatively, a refrigeration system can be used for providing cooling in summer and heating in winter. Such systems have been built and are available now. Industrial air conditioning is largely responsible for the growth of modern electronic, pharmaceutical, chemical industries etc. Most of the present day air conditioning systems use either a vapor compression refrigeration system or a vapor absorption refrigeration system. The capacities vary from few kilowatts to megawatts.

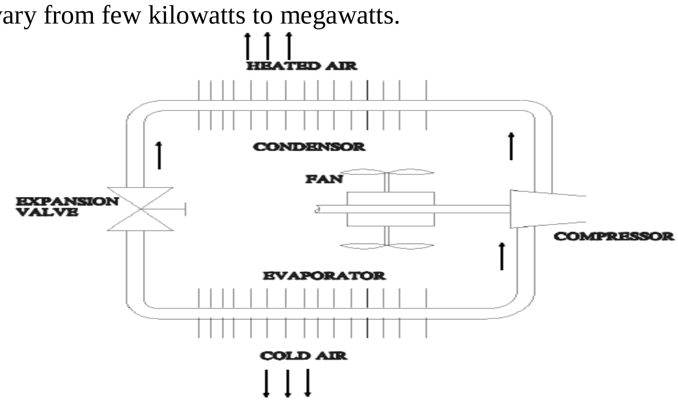

Fig-1: Schematic of a basic vapor compression refrigeration system[1]

As shown in the figure 1 the basic system consists of an evaporator, compressor, condenser and an expansion valve. The refrigeration effect is obtained in the cold region as heat is extracted by the vaporization of refrigerant in the evaporator. The refrigerant vapor from the evaporator is compressed in the compressor to a high pressure at which its saturation temperature is greater than the ambient or any other heat sink[5]. Hence when the high pressure, high temperature refrigerant flows through the condenser, condensation of the vapor into liquid takes place by heat rejection to the heat sink. To complete the cycle, the high pressure liquid is made to flow through an expansion valve. In the expansion valve the pressure and temperature of the refrigerant decrease. This low pressure and low temperature refrigerant vapour evaporates in the evaporator taking heat from the cold region. It should be observed that the system operates on a closed cycle. The system requires input in the form of mechanical work. It extracts heat from a cold space and rejects heat to a high temperature heat sink. Energy of food is converted into chemical energy for functioning of brain, lungs, heart and other organs and this energy is ultimately rejected to the surroundings. Also the internal organs require a temperature close to $35^{o}C$ for their efficient operation, and regulatory mechanisms of human body maintain this temperature by rejecting appropriate amount of heat[2]. Human beings do not feel comfortable if some extra effort is required by the body to reject this energy. The air temperature, humidity and velocity at which human body does not have to take any extra action, is called comfort condition. The residences, offices, shopping centers, stores, large buildings, theaters, auditorium etc. all have slightly different requirements and require different design. The required cooling capacities also vary widely depending upon the application. The factory assembled from a fraction of a ton (TR) to about 2 TR. These systems use a vapor compression refrigeration system with a sealed compressor and forced convection type evaporators and condensers. Figure 1.2 shows the schematic of a widow type room air conditioner. In this type all the components are housed in a single outer (unit) are housed in a separate casing and is kept away from the indoor unit consisting of the evaporator, blower, filter etc. The outdoor and indoor units are connected by refrigerant piping. For medium sized buildings factory assembled package units are available, while for very large buildings a central air conditioning system is used. This is specially so for the operation theatres and intensive care units. In these places no part of the room air is re-circulated after conditioning by A/C system. In other places up to 90% of the cold room air is re-circulated and 10% outdoor fresh air is taken to meet the ventilation requirement of persons. In hospitals all the room air is thrown out and 100% fresh air is taken into the A/C system. Since, outdoor air may be at $45^{o}C$ compared to $25^{o}C$ of the room air, the air-conditioning load becomes very large. The humidity load also increases on this account. Operation theatres require special attention in prevention of spores, viruses, bacteria and contaminants this purpose. Restaurants, theatres and other places of amusement require air-conditioning for the comfort of patrons. All places where, a large number of people assemble should have sufficient supply of fresh air to dilute CO_2 and body odours emitted by persons[9]. In addition, people dissipate large quantities of heat that has to be removed by air-conditioning for the comfort of persons. These places have wide variation in air-conditioning load throughout the day. These have large number of persons, which add a lot of water vapors by respiration and perspiration. The food cooked and consumed also adds water vapors. This vapor has to be removed by air-conditioning plant. Hence, these buildings have large latent heat loads. Infiltration of warm outdoor is also large since a number of persons enter and leave the building leading to entry of outdoor air with every door opening. Ventilation requirement is also very large. Syndrome is very common in poorly designed air conditioned buildings due to in advent the feeling of nausea, headache, eye and throat irritation and the general feeling of being uncomfortable with the indoor environment. In developed countries this is leading to litigation also. In the earlier systems little attention was paid to energy conservation, since fuels were abundant and inexpensive. The energy crisis in early seventies, lead to a review of basic principles and increased interest in energy optimization. The concept of low initial cost with no regard to operating cost has become obsolete now. Approaches, concepts and thermodynamic cycles, which were considered impractical at one time, are receiving longer period of stay and thereby increases the sales. Supermarkets have frozen food section, refrigerated food section, dairy and brewage section, all of them requiring different temperatures[13]. The refrigeration system has to cater to different temperatures, apart from air-conditioning. These places also have a wide variation in daily loads depending upon busy and lean hours, and holidays. Large commercial buildings are a world of their own; they have their own shopping Centre, recreation Centre, gymnasium swimming pool etc. Offices have very high density of persons during office hours and no occupancy during off time[18]. These buildings require integrated concept with optimum utilization of resources and services. These have security aspects, fire protection, emergency services, optimum utilization of energy all built-in. Modern buildings of this type are called intelligent buildings where air-conditioning requires large amount of energy and hence is the major focus.

Since persons have to spend a major part of their time within the building, without much exposure to outdoors, the concept of Indoor Air Quality (IAQ) has become very important. There are a large number of pollutants that are emitted by the materials used in the construction of buildings and brought into the buildings.

IAQ addresses to these issues and gives recommendation for their reduction to safe limits. Sick building serious considerations now. Earlier, the index of performance used to be first law efficiency, no a recovery systems, alternate refrigerants and mixtures of refrigerants are being proposed to optimize energy use. Large-scale applications of air-conditioning in vast office and industrial complexes and increased awareness of comfort and indoor air quality have lead to challenges in system design and simulations. Developments in electronics, controls and computers have made refrigeration and air-conditioning a high-technology industry.

II. VAPOR COMPRESSION REFRIGERATION SYSTEM

Description

Compression refrigeration cycles take advantage of the fact that highly compressed fluids at ascertain temperature tend to get colder when they are allowed to expand. If the pressure change is high enough, then the compressed gas will be hotter than our source of cooling (outside air, for instance) and the expanded gas will be cooler than our desired cold temperature. In this case, fluid is used to cool a low temperature environment and reject the heat to a high temperature environment[3]. Vapor compression refrigeration cycles have two advantages. First, a large amount of thermal energy is required to change a liquid to a vapor, and therefore a lot of heat can be removed from the air-conditioned space. Second, the isothermal nature of the vaporization allows extraction of heat without raising the temperature of the working fluid to the temperature of whatever is being cooled. This means that the heat transfer rate remains high, because the closer the working fluid temperature approaches that of the surroundings, the lower the rate of heat transfer. The refrigeration cycle is shown in Figure 2 and can be broken down into the following stages:

1–2 Low-pressure liquid refrigerant in the evaporator absorbs heat from its surroundings, usually air, water or some other process liquid. During this process it changes its state from a liquid to a gas, and at the evaporator exit is slightly superheated.

2 – 3 The superheated vapors enter the compressor where its pressure is raised. The temperature will also increase, because a proportion of the energy put into the compression process is transferred to the refrigerant.

3– 4 The high pressure superheated gas passes from the compressor into the condenser. The initial part of the cooling process (3-3) de-superheats the gas before it is then back into liquid (3a-3b). The cooling for this process is usually achieved by using air or water. A further reduction in temperature happens in the pipe work and liquid receiver (3b-4), so that the refrigerant liquid is sub-cooled as it enters the expansion device.

4 - 1 The high-pressure sub-cooled liquid passes through the expansion device, which both reduces its pressure and controls the flow into the evaporator.

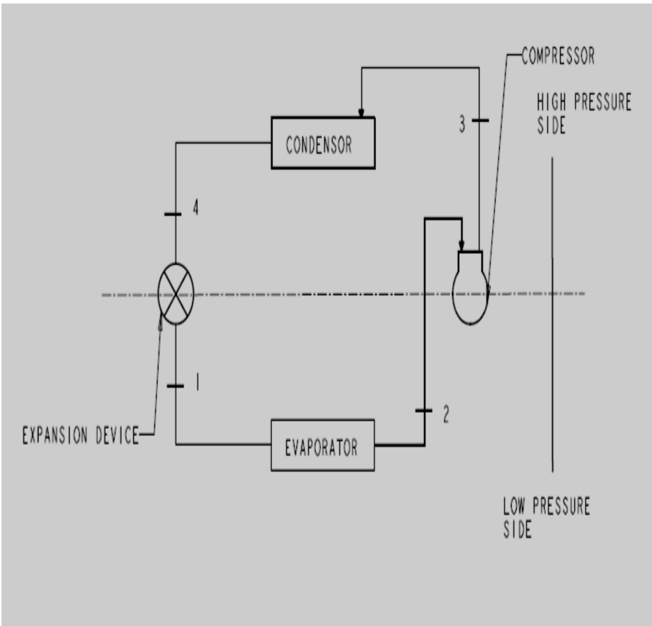

Fig- 2: Schematic representation of the vapour compression refrigeration cycle[2]

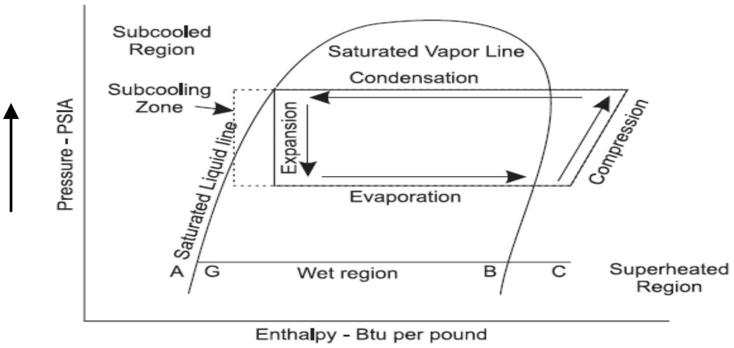

Fig-3: Schematic representation of the refrigeration cycle including pressure change.

Types of refrigerant used in vapor compression systems

A variety of refrigerants are used in vapor compression systems. The required cooling temperature largely determines the choice of fluid. Commonly used refrigerants are in the family of chlorinated fluorocarbons (CFCs, also called Freons): R-11, R-12, R-21, R-22 and R-502. The properties of these refrigerants are summarized in Table 1 and the performance of these refrigerants is given in Table1 below.

Refrigerant	Boiling point (^oC)	Freezing point (^oC)	Vapor Press. (kPa)	Vapor Vol. m^3/Kg	Enthalpy	
					Liquid kJ/kg	Vapor kJ/kg
R- 11	-23.82	-111	25.73	0.6117	191.4	385.4
R-12	-29.79	-158	219.28	0.0770	190.7	347.9
R- 22	-40.76	-160	354.74	0.0651	188.5	400.8
R- 502	-45.4	-----	414.3	0.0423	188.8	342.3
R-7 (Ammonia)	-33.3	-77.7	289.93	0.4194	808.7	487.7

Table- 1 : Properties of commonly used refrigerants[2]

Refrigerant	Evaporating Press (kPa)	Condensing Press (kPa)	Pressure Ratio	Vapor Enthalpy (kJ/kg)	COP
R-11	20.4	125.5	6.15	155.4	5.03
R-12	182.7	744.6	4.08	116.3	4.70
R-22	295.8	1192.1	4.03	162.8	4.66
R-502	349.6	1308.6	3.74	106.2	4.37
R-717	236.5	1166.5	4.93	103.4	4.78

Table-2: Performance of commonly used refrigerants [2]

III. RESIDENTIAL AIR CONDITIONING SYSTEM

Air Conditioner Components

Compressor

The purpose of the compressor is to increase the working pressure of the refrigerant. Compressors fall into two general categories: positive displacement, which increase the pressure of the vapor by reducing the volume, and dynamic, which convert angular momentum into a pressure rise and transfer it to the vapour[21]. Scroll type, positive displacement compressors which dominate the residential compressor market were considered for this study. The amount of specific work done by an ideal compressor can be found by the energy equation[5]:

$$w_{s,com} = (h_{2s} - h_1) \qquad \text{--------------- 1)}$$

where: h_1= refrigerant enthalpy entering compressor

h_{2s} = refrigerant enthalpy isentropic compressor

For a non-ideal compressor, the actual amount of work required depends on the efficiency.

$$w_{a,com} = \frac{w_{s,com}}{\eta_c} = (h_{2s} - h_1) \text{ --------------- } 2)$$

Where:

$$\eta_c = \text{compressor thermal efficiency}$$

For a scroll type compressor, Klein has determined the thermal efficiency is related to the reduced pressure and reduced temperature with the following equation.

$$\eta_c = -60.25 - 3.814P_{rat} - 0.281P_{rat}^2 + 111.3T_{rat} - 50.31T_{rat}^2 + 3.061P_{rat}\ T_{rat} \text{ ---------- } 3)$$

$$\text{Where } P_{rat} = \frac{P_{sat,cond}}{P_{sat,evap}} \text{ ------------------ } 4)$$

$$T_{rat} = \frac{T_{sat,cond}}{T_{sat,evap}} \text{ ---------------- } 5)$$

The author only considers the saturated sections of the heat exchangers. Therefore, the coefficients in the compressor efficiency correlation are based on the saturated temperatures rather than the actual inlet and outlet temperatures to the compressor. Since pressure drops are included in the condenser model, the compressor efficiency is based on the inlet saturation temperature and pressure in the condenser and the outlet saturation temperature and pressure from the evaporator[8].

It is important to consider the volumetric efficiency in addition to the thermal efficiency. The volumetric efficiency is the ratio of the mass of vapor that is compressed to the mass of vapor that could be ideally compressed if the intake volume were equal to the piston displacement and filled with evaporator exit state vapor. The volumetric efficiency can be found by

$$\eta_v = 1 - R_{cv,pd}\left(\frac{v_1}{v_2} - 1\right) \text{ --------------- } 6)$$

Where: η_v = compressor volumetric efficiency

$R_{cv,pd}$ = ratio piston/ cylinder of clearance volume to swept displacement

v_1 = specific volume at entering the compressor

$v_2 = specific\ volume\ compressor\ out$

The volumetric efficiency is used to determine the mass flow rate of the refrigerant through the compressor for a given compressor size by the following expression,

$$\dot{m} = \frac{\eta_v PD}{v_2} \text{ -----------------------} 7)$$

where: PD = piston displacement

Condenser : The condenser is a heat exchanger that rejects heat from the refrigerant to the outside air. Heat exchangers come in many configurations, but finned-tube heat exchangers are most common for residential air conditioning applications. Refrigerant flows through the tubes and a fan forces air between the fins and over the tubes [9]. The heat exchangers used in this study will be the plate finned-tube type as shown in Fig. 4.

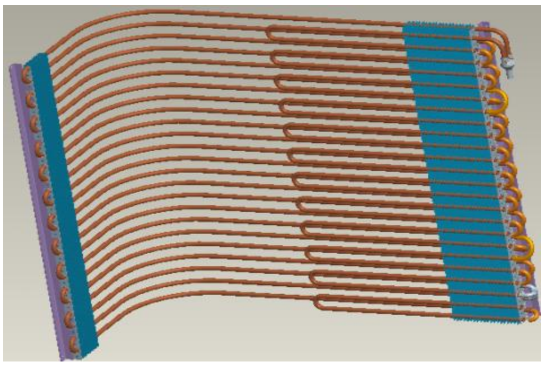

Fig- 4: Typical Plate Finned-Tube Heat Exchanger

When the refrigerant leaves the compressor, it enters the condenser as a superheated vapor and leaves as a sub-cooled liquid. The condenser can be separated into three sections: superheated, saturated, and sub-cooled. The specific heat rejected from each section can be found by evaluating the refrigerant enthalpies at the inlets and outlets.

$$q_{con,sh} = h_2 - h_{2a} \text{ ---------------------} 8)$$

Condenser Fan

Because natural convection will not produce sufficient airflow and heat transfer over a reasonably sized condenser, a fan must be employed to keep the air moving [10].

Although the compressor uses the majority of the power consumed by the system, the fan power must also be considered. The power required by the fan is directly related to the air pressure drop across the condenser and the volume flow rate of the air.

$$\dot{W_{f,con}} = \frac{V_{a,con}\,\Delta P_{a,con}\,A_{fr,con}}{\eta_{fan,con}} \text{ --------------------9)}$$

$$h_3 = h_4 \qquad \text{---------------------------10)}$$

where: V_a = air velocity

$\Delta P_{a,con}$ = air-side pressure drop

$A_{fr,con}$ = frontal area

$\eta_{fan,con}$ = fan efficiency

The isentropic efficiency of the combined fan and motor is taken to be 65%.

Expansion Valve : The expansion valve is used to control the refrigerant flow through the system. Under normal operating conditions, the thermo-static expansion valve opens and closes to maintain a fixed superheat exiting the evaporator[13]. In this study, that superheat will be held at 10^{o} F. Because the expansion valve is designed to pass a certain volume of refrigerant, it cannot function properly if the refrigerant is not completely condensed. The vapor refrigerant backs up behind the valve and the condenser pressure increases until the refrigerant vapor is condensed. When this happens, the expansion valve cannot regulate the refrigerant superheat exiting the evaporator. Under this condition, it converts from maintaining superheat to maintaining a saturated liquid leaving the condenser. The energy equation shows that the enthalpy is constant across the expansion valve.

Evaporator : The purpose of the evaporator is to transfer heat from the room air making the air cooler and less humid[17]. Because the refrigerant enters the evaporator in as a liquid-vapor mix, it is only divided into saturated and superheated sections. The analysis for the evaporator is nearly identical to that of the condenser, but some considerations must be made for the dehumidification process. To keep the evaporator model simple, the coil is assumed to be dry, so the air-side heat transfer coefficient is not affected, but the specific heat is corrected to account for condensation. Because the air flowing over the evaporator is cooled below the wet bulb temperature, some of the heat rejected by the air results in condensing water out of the air rather than lowering the temperature. The total enthalpy change of the air is the sum of the enthalpy change due to temperature drop, or sensible heat, and the enthalpy change due to condensation, or latent heat.

$$\Delta h_{tot} = \Delta h_{sens} + \Delta h_{lat} \text{ ----------------- 11)}$$

As shown in Figure 5 using the specific heat for dry air will result in exit temperatures that are too low. By using an effective specific heat, a more accurate exit temperature can be obtained without the complications associated with using an air-water mixture.

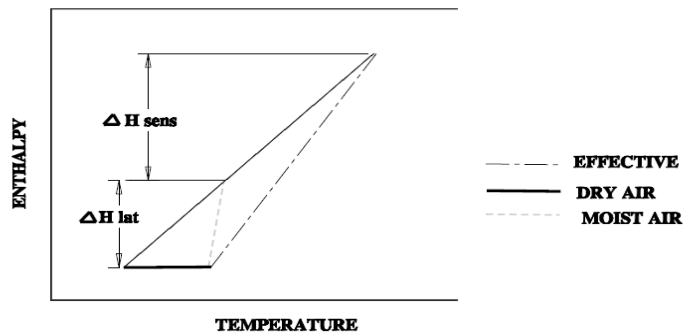

Fig- 5: Effective specific heat

Evaporator Fan

Because the evaporator is not the focus of this study, introducing wet coils would introduce unwelcome complications. In addition to affecting the heat transfer, wet coils also affect the air-side pressure drop. Although there are correlations available to find the pressure drop over wet coils, this is not the only issue[16]. After the air flows over the evaporator, it enters a series of ducts that return it to the inside living space. Because the power required depends on the losses in the ducts which will change from case to case, the default used by the Air-conditioning and Refrigeration Institute test standard of 365 W per 1000 cfm of air will be used.

Coefficient of Performance

The coefficient of performance (COP) measures the efficiency of the entire system. It is the ratio of the heat absorbed by the evaporator to the amount of electrical energy used by the mechanical components, i.e. the compressor and two fans[21].

$$\text{COP} = \frac{\dot{Q}_e}{\dot{W}_{con} + \dot{W}_{f,con} + \dot{W}_{f,evap}} \quad \text{---------------- 12)}$$

Seasonal COP

The American Refrigeration Institute has determined that the frequency distribution of temperatures over the summer cooling season is roughly the same across the country[21]. However, in warmer, southern climates, there are more "cooling load hours", which are defined as the hours when the temperature is above 65 F, per year than in cooler climates. In Atlanta, for example, the number of cooling load hours is approximately 1300 hours per year, while it is only about 700 hours per year in sea level and OH. Of these hours, the outside temperature will be between 80 F and 84 F approximately 16.1% of the cooling season in either city.

Bin Number	Temperature Range (F)	Representative Temperature (F)	Fraction of Total Temperature Hours
1	65-69	67	0.214
2	70-74	72	0.231
3	75-79	77	0.216
4	80-84	82	0.161
5	85-89	87	0.104
6	90-94	92	0.052
7	95-99	97	0.018
8	100-104	102	0.004

Table- 5. Distribution of Cooling Load Hours[21]

The COP changes with the outside air temperature and the overall COP, or seasonal COP, for an air conditioner depends on the temperatures at which the appliance runs over an entire year. According to the ANSI/ASHRAE standard, the seasonal COP for a single speed, single compressor unit is found.

The adjusted evaporator capacity and adjusted electrical power demand are based on the cooling load factor and the fraction of total temperature hours.

Evaporator Fan :Because the evaporator is not the focus of this study, introducing wet coils would introduce unwelcome complications. In addition to affecting the heat transfer, wet coils also affect the air-side pressure drop. Although there are correlations available to find the pressure drop over wet coils, this is not the only issue[23]. After the air flows over the evaporator, it enters a series of ducts that return it to the inside living space. Because the power required depends on the losses in the ducts which will change from case to case, the default used by the Air-conditioning and Refrigeration Institute test standard of 365 W per 1000 cfm of air will be used.

IV. HEAT EXCHANGERS

Geometry

The "tubes per circuit" is the number of parallel passages the refrigerant mass flow rate is divided among. If the mass flow rate is 100 lbm/hr and there is one tube per circuit, 100 lbm/ hr of refrigerant will pass through that tube. If 2 tubes per circuit, then 50 lbm/hr of refrigerant will flow through each tube. The number of parallel circuits is used to determine the number of tubes in each row. The number of rows refers to the number of tube rows in the direction normal to air flow. If the number of parallel circuits is set to 12, and the number of tubes per circuit is 2, then there will be a total of 24 tubes in each row.

The fin pitch is the number of fins per unit length along the axial direction of the tubes. These parameters and the overall dimensions of the heat exchanger are illustrated in figure 5 and figure 6. The model used to determine the air-side heat transfer coefficient depends on the layout of the tubes, but not on the temperature of the refrigerant which would be affected by circuiting. The only factors pertinent to the refrigerant side models affected by the circuiting are the mass flow rate in each tube and the length associated with each circuit. The layout in Figure 5 meets the requirements of the models and is easy to conceptualize.

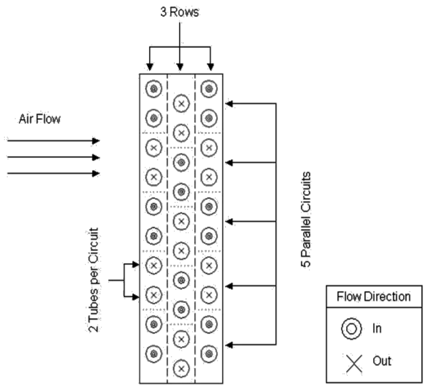

Fig-6: General Heat Exchanger Dimensions[23]

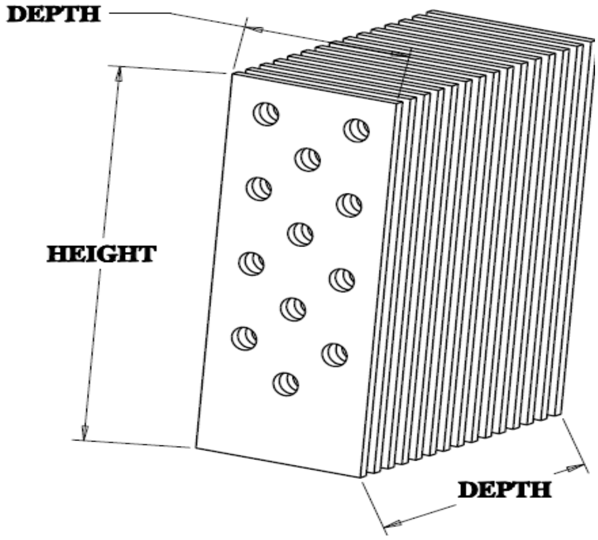

Fig-7: Layout of Heat Exchanger Geometry Parameters[23]

NTU-Effectiveness Relations

For any heat exchanger, the total heat rejected from the hot fluid, in this case, refrigerant, to the cold fluid, air, is dependent on the heat exchanger effectiveness and the heat capacity of each fluid.

$$Q = \varepsilon C_{min} (T_{h,i} - T_{c,i}) \text{ -------------------13)}$$

where: = effectiveness
C_{min} = smaller of heat capacities C_h and C_c
$T_{h,i}$ = inlet temperature of hot fluid
$T_{c,i}$ = inlet temperature of cold fluid

The heat capacity, C, the extensive equivalent of the specific heat, determines the

amount of heat a substance absorbs or rejects per unit temperature change.

$$C = mc_p \text{ --------------------------14)}$$

Where m = mass
c_p = specific heat

The amount of air flowing over each section of the condenser is assumed to be proportional to the tube length associated with that section.
The effectiveness is the ratio of the actual amount of heat transferred to the maximum possible amount of heat transferred.

V. RESULTS AND DISCUSSIONS

Effect of FIN tube heat condenser with 1.5 row coil on the cooling of 1.5TR split air conditioner.

Voltage(V)	230.10	230.09	230.10	230.09	230.08	230.09	230.10	230.09
Current(A)	7.54	7.54	7.54	7.55	7.55	7.55	7.52	7.54
Power(W)	1718.00	1711.00	1712.00	1715.0	1713.00	1713.00	1718.0	1714.29
Cooling Capacity (BTU)	17101.3	17128.2	17057.3	17183.6	17155.7	17157.1	17112.2	17127.9
C.O.P	2.93	2.93	2.92	2.95	2.90	2.94	2.97	2.94
Suction	13.70	13.80	14.00	13.80	13.70	13.70	13.70	13.77
Discharge	85.70	85.60	85.80	85.80	85.70	85.70	85.60	85.70
Liquid	41.40	41.50	41.60	41.70	41.70	41.50	41.50	41.56

Table- 6: show the observation values being measured including voltage, current, power, cooling capacity, suction temperature, discharge temperature, and liquid temperature.

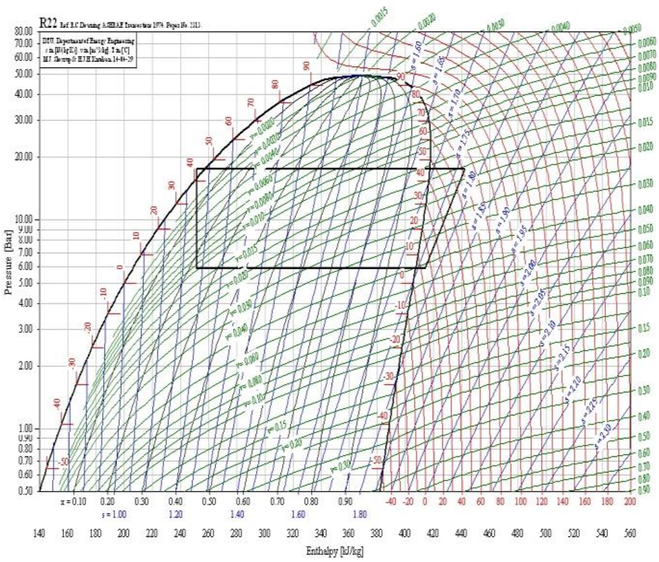

Fig-8: P-H diagram **o**f FIN tube heat condenser with 1.5 row coil on the cooling of 1.5TR split air conditioner.

Effect of FIN tube heat condenser with 2.0 row coil on the cooling of 1.5TR split air conditioner.

The following observation were noticed for FIN tube heat condenser with 2.0 row coil (755+755):

Voltage	230.08	230.08	230.08	230.08	230.08	230.08	230.09	230.08
Current	7.52	7.53	7.52	7.52	7.52	7.52	7.50	7.52
Power	1695.00	1692.00	1688.00	1689.00	1688.00	1690.00	1686.00	1689.71
Cooling Capacity	17482.70	17377.00	17364.27	17445.11	17455.21	17429.19	17495.34	17435.55
C.O.P	3.02	3.01	3.01	2.99	3.00	2.99	3.02	3.01
Suction	9.40	9.50	10.00	9.90	10.20	10.00	10.00	9.86
Discharge	82.90	82.90	82.90	82.80	82.70	82.80	82.60	82.80
Liquid	41.40	41.30	41.10	41.20	41.20	41.20	41.10	41.21

Table- 7: Effect of FIN tube heat condenser with 2.0 row coil on the cooling of 1.5TR split air conditioner.

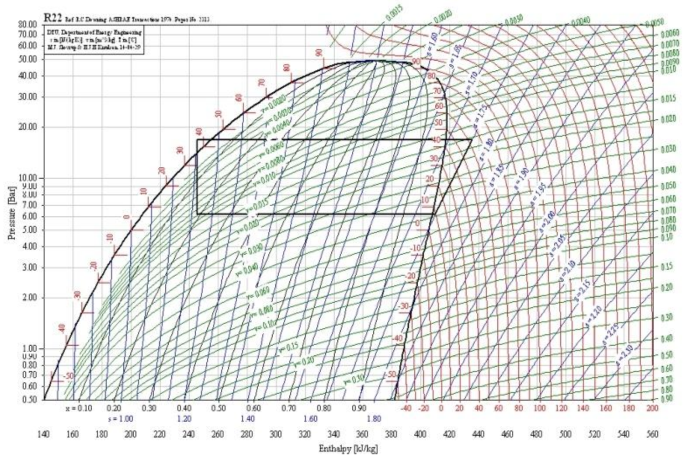

Fig-9: P-H diagram of FIN tube heat condenser with 2.0 row coil on the cooling of 1.5TR split air conditioner.

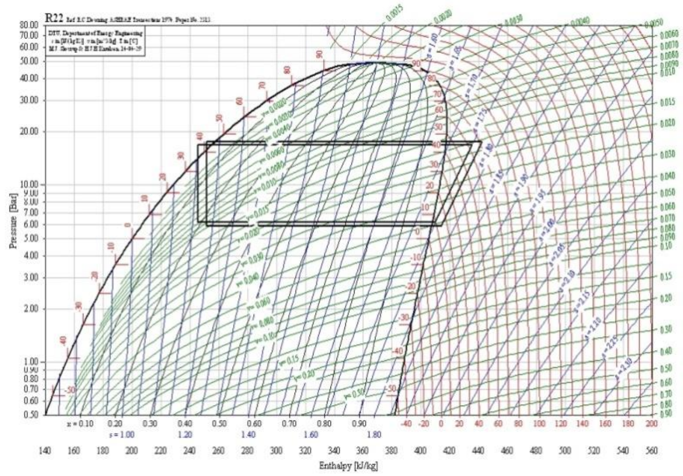

Fig-10: P-H diagram of FIN tube heat condenser with 1.5 row and 2.0 row coil on the cooling of 1.5TR split air conditioner.

Comparison of Current for 1.5 row and 2 row condenser with respect to time.

The following observations were noticed while comparing the current for 1.5 row and 2 row condenser:

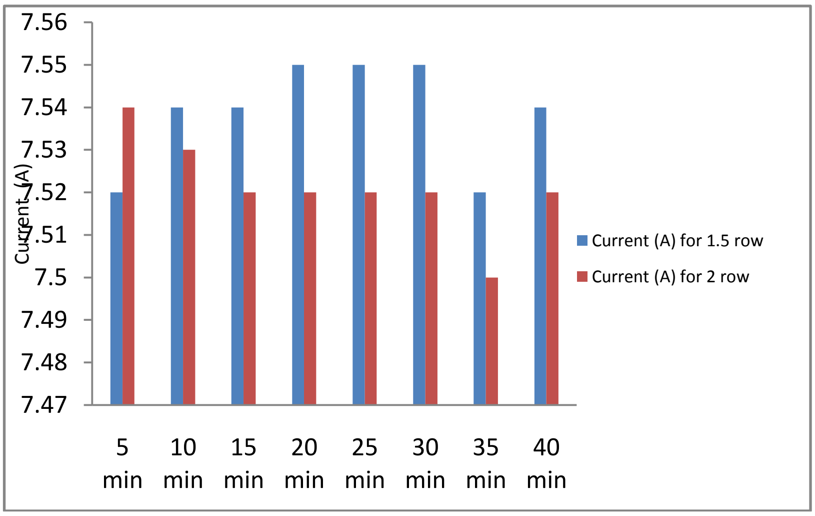

Fig-11: Graphical presentation of the current for 1.5 row and 2 row condenser

From the figure 11 it can be observed that after interval of 5 minutes current in the case of 2 row coil reduced by 4.5 percent. Maximum 7.55 A current observed in the 1.5 row coil. Minimum 7.5 A current observed in the 1.5 row coil. Time interval 15 to 30 minutes constant current observed in the case of 2 row coil.

Comparison of Power for 1.5 row and 2 row condenser with respect to time.

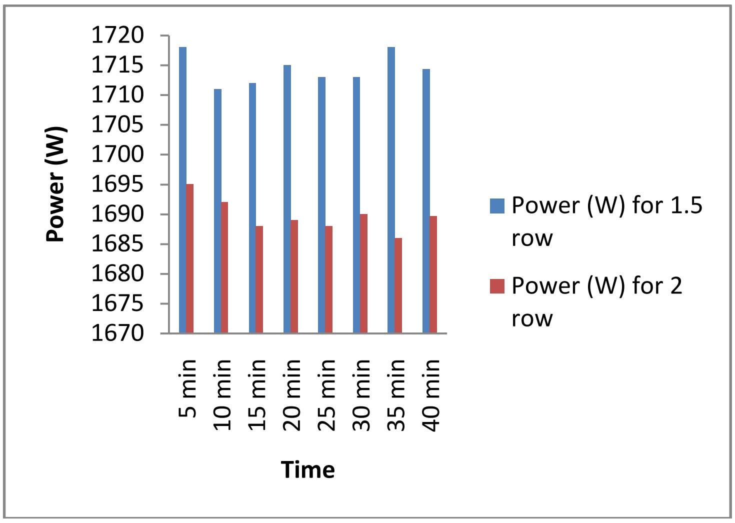

Fig-12 : Graphical presentation of the power for 1.5 row and 2 row condenser

From the figure 12 it can be seen that the low input power is observed in the case of 2 row coil and more power is consumed in the case of 1.f row coil. Maximum power is consumed between 1515 to 1520 watt in the case of 1.5 row coil and minimum 1685 watt is observed in the 2 row coil. In 1.5 row coil input power is observed between 1710 to 1720 watts. Reason for reduction of Input Power: When the condenser size increased, the condensing pressure goes down, the discharge pressure also gone down correspondingly. By studying the p-h diagram of the refrigerant we came to know that the lower discharge pressure would reduce the compression ratio of the compressor so less work would be required at compressor's end. Hence the less input power is required.

Comparison of Cooling Capacity for 1.5 row and 2 row condenser with respect to time.

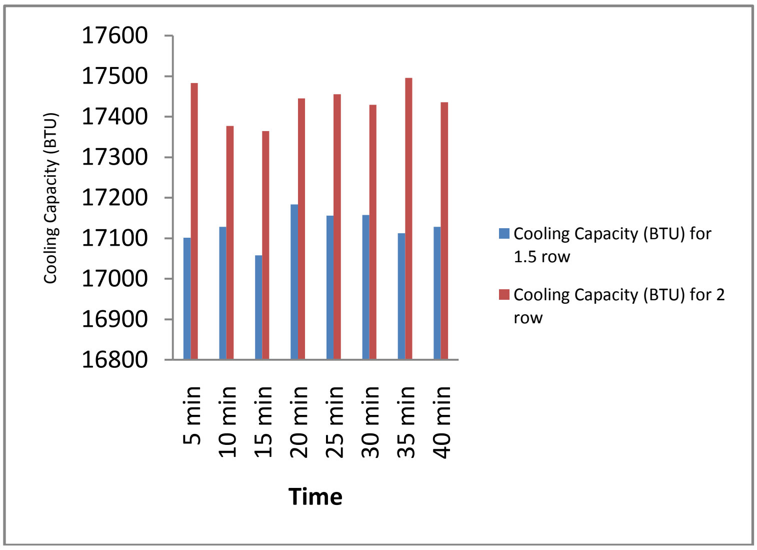

Fig-13: Graphical presentation of Cooling Capacity for 1.5 row and 2 row condenser with respect to time.

From the figure 13, it can be seen that minimum 17050 btu/hr observed in the case of 1.5 row coil. Maximum 17500 btu/hr. is observed in the case of 2.0 coil. In all observation higher cooling capacity is achieved in the case of 2.0 row coil and low capacity is observed in the case of 1.5 row coil. Reason for Capacity Increase: Increasing the size of condenser will ensure more liquid after the expansion valve due to more subcooling. Increased surface area of condenser will offer more heat transfer to the surroundings. The quality of mixture entering the Evaporator will be improved so refrigeration effect will be improved due to the availability of more latent heat transfer capacity of the liquid refrigerant.

Comparison of C.O.P with 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger.

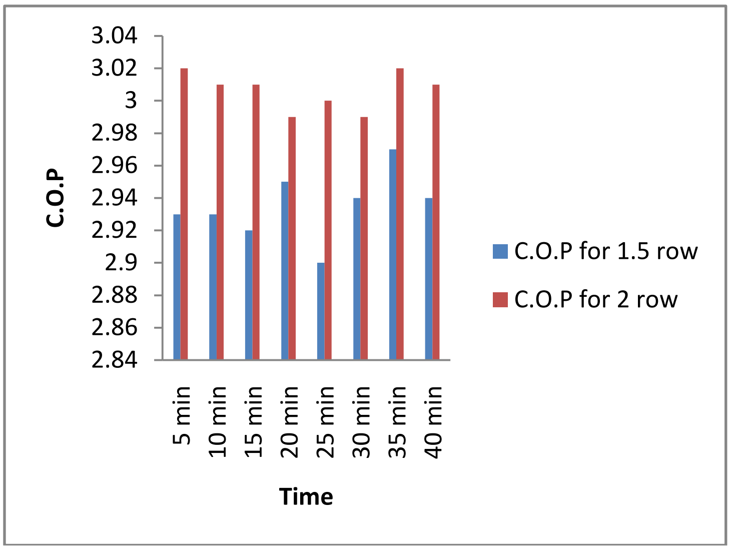

Fig-14: Graphical representation of C.O.P with 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger.

From figure 14, it can be seen that in all the observation COP is higher in the case of 2 row coil and low in the case of 1.5 row coil. Minimum COP 2.9 in 1.5 row coil and maximum 3.02 COP in the case of 2 row coil is observed .Reason for increased COP in 2 row : In 2 row area of heat exchanger is increased because of this more heat is transferred from the refrigerant because subcooling increases.

Comparison of Suction (T°) with 1.5 Row and 2.0 Row Fin Tube Condenser heat exchanger.

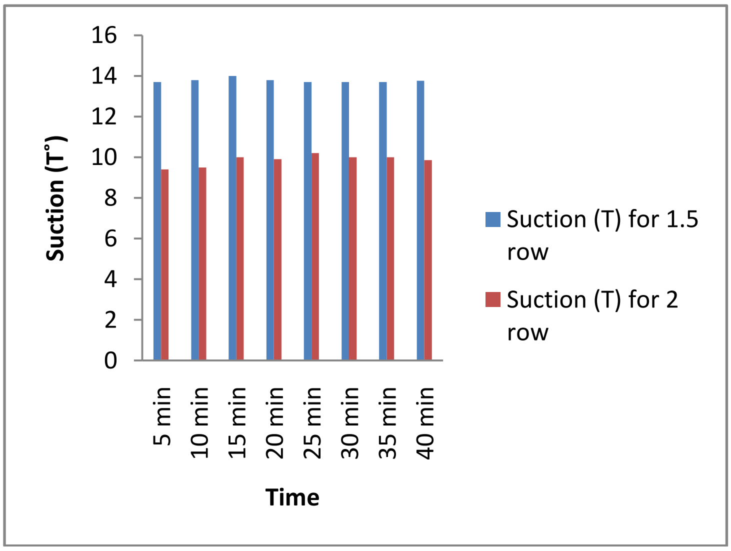

Fig-15: Graphical representation of Suction temp. of 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger.

From the figure 15 it can be observed that in all the observations suction temperature is in lower side in the case of 2.0 row coil and observed higher in the case of 1.5 row coil. In 2 row coil suction temperature observed

between 9.5 to 10 degree. Maximum suction observed 13.9 degree in 1.5 row coil. In 1.5 row coil suction temperature observed 13.9 to 14 degree.
Reason for increased Suction Pressure: Increasing the size of condenser will eliminate the requirement of long capillary size and by reducing the capillary in the same system where the comparatively small condenser being used, the suction pressure will be improved.

Comparison of Discharge (T°) with 1.5 Row and 2.0 Row Fin Tube Condenser heat Exchanger

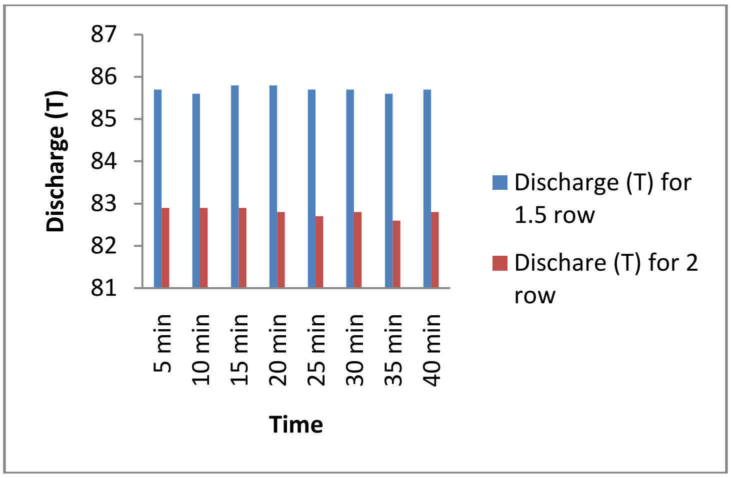

Fig-16: Graphical representation of C.O.P with 1.5 Row and 2.0 Row Fin Tube Condenser heat exchanger.

From the figure 16, it can be seen that discharge temperature is on higher side in all observation in 1.5 row coil and low in the 2 row coil. Discharge temperature observed between 82 to 83 degree in the 2 row coil and 85 to 86 in the 1.5 row coil. Reason for the reduction in the discharge pressure: As mentioned above, the increased size of condenser will reduce the compression ratio of the compressor so the outlet discharge pressure will be less.

Comparison of liquid temperature with 1.5 Row and 2.0 Row Fin Tube Condenser heat exchanger

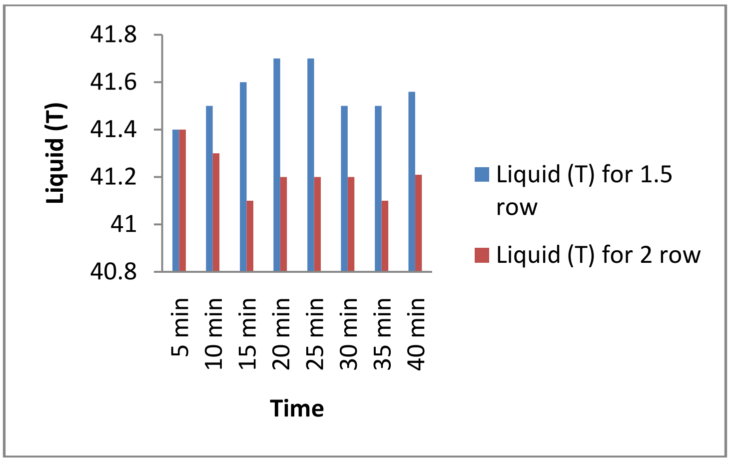

Fig-17: Graphical representation of liquid with 1.5 Row and 2.0 Row Fin Tube Condenser heat exchanger.

From the figure 17 it can be seen that the liquid temperature is same for 1.5 row and 2 row coil. Minimum 41.1 degree liquid temperature observed in the 2 row coil. Maximum 41.7 degree temperature is observed in the 1.5 row coil.

Reason for the improvement in liquid line temperature: Increased condenser size will offer more heat transfer area to the refrigerant. The temperature of the liquid will be reduced as the condensing temperature is also low due to less discharge pressure.

VI. CONCLUSION

In this investigation an accurate condenser model was developed and integrated into an air-conditioning system. A base condenser model was chosen and design conditions were established at 50^o. The operating parameters of condenser subcool and air face velocity were examined over a wide range of ambient conditions to determine their effects on the COP. From this study is concluded that the increased area of condenser ensures more heat transfers in condenser which increases subcooling i.e. input power goes on lower side. Increasing the size of condenser will ensure more liquid after the expansion valve due to more subcooling. Increased surface area of condenser will offer more heat transfer to the surroundings. The quality of mixture entering the Evaporator will be improved so refrigeration effect will be improved due to the availability of more latent heat transfer capacity of the liquid refrigerant. Increased condenser size will offer more heat transfer area to the refrigerant. The temperature of the liquid will be reduced as the condensing temperature is also low due to less discharge pressure. When the condenser size will be increase, the condensing pressure will go down, the discharge pressure will also go down correspondingly. By analysing the p-h diagram of the refrigerant it is concluded that the lower discharge pressure reduces the compression ratio of the compressor so less work will be required at compressor's end. Increasing the size of condenser will eliminate the requirement of long capillary size and by reducing the capillary in the same system where the comparatively small condenser being used, the suction pressure will be improved. As mentioned above, the increased size of condenser will reduce the compression ratio of the compressor so the outlet discharge pressure will be less.

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